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Effects on Diagnostic Parameters After Removing Additional Synchronous Gear Meshes

Harry J. Decker
U.S. Army Research Laboratory, Glenn Research Center, Cleveland, Ohio

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Harry J. Decker
U.S. Army Research Laboratory
Cleveland, Ohio 44135
Harry.J.Decker@grc.nasa.gov

Abstract: Gear cracks are typically difficult to diagnose with sufficient time before catastrophic damage occurs. Significant damage must be present before algorithms appear to be able to detect the damage. Frequently there are multiple gear meshes on a single shaft. Since they are all synchronous with the shaft frequency, the commonly used synchronous averaging technique is ineffective in removing other gear mesh effects. Carefully applying a filter to these extraneous gear mesh frequencies can reduce the overall vibration signal and increase the accuracy of commonly used vibration metrics. The vibration signals from three seeded fault tests were analyzed using this filtering procedure. Both the filtered and unfiltered vibration signals were then analyzed using commonly used fault detection metrics and compared. The tests were conducted on aerospace quality spur gears in a test rig. The tests were conducted at speeds ranging from 2500 to 5000 revolutions per minute and torques from 184 to 228 percent of design load. The inability to detect these cracks with high confidence results from the high loading which is causing fast fracture as opposed to stable crack growth. The results indicate that these techniques do not currently produce an indication of damage that significantly exceeds experimental scatter.

Introduction: There is considerable work being performed in Health and Usage Monitoring Systems (HUMS) to reduce maintenance of mechanical components such as gearboxes and to increase vehicle safety. Health and Usage Monitoring can be classified into two major areas: diagnostics and prognostics. Diagnostics deals with the consistent and accurate detection of damage, while prognostics include both damage estimation and estimating the remaining useful life.

A major concern of current HUMS systems is their reliability. A recent report proposes that the current fault detection rate of a vibration-based system is 60 percent. A false alarm is typically generated every hundred hours. [1, 2]

Since 1988, NASA Glenn Research Center has been working on improving gear damage detection using vibration monitoring. Most of the effort has focused on pitting and other surface distress failures. Later, the testing expanded into oil debris monitoring-based HUMS as well as vibration based crack detection and propagation. Gear cracks, although potentially more catastrophic, are much less common, thus more difficult to study.

Theory: Many different techniques have been proposed to detect damage in mechanical power transmissions. These methods include vibration, oil debris detection, chemical element detection, and acoustic emission. The focus of this paper is vibration analysis.

One of the processes that virtually all of the existing diagnostic techniques require is synchronous averaging. Synchronous averaging has two desirable effects: (1) it reduces the effects of items in the vibration signal that are not synchronous with shaft and mesh frequencies and (2) because of this, the signal to noise ratio is increased. The averaging technique typically used is synchronous with shaft revolution.

When more than one gear is on a shaft there is the likelihood that this other gear will add spectral components that will corrupt, or even totally mask, the vibration signal of interest. One technique to remove the undesired components is to filter them out of the signal. This paper examines the effect of having multiple synchronous gear meshes. In this paper, the filtering occurred in the frequency domain. The time domain signal is transformed into the frequency domain using the Fast Fourier Transform, the offending frequencies minimized and the resultant signal is transformed back into the time domain. Care must be exercised to ensure that only those gear mesh frequencies (and harmonics) that do not occupy the same frequency bin are altered.

The resultant averaged signals are then analyzed using the following metrics: root mean square (RMS), Kurtosis (Kurt), Crest Factor (CF), Energy Ratio, FM0, FM4, M6A, NA4, NA4*, NB4 and NB4*. The references for these metrics can be found on the World Wide Web. [3] The details for these metrics are beyond the scope of this paper.

Experiment Configuration:

Facility Description: A spur gear fatigue test stand at the NASA Glenn Research Center in Cleveland, Ohio, is used to perform the testing. This facility, shown in Figure 1, is used to study of effects of gear tooth design, gear materials, and lubrication on the fatigue lives of aerospace quality gears. The test stand operates using the closed loop torque regeneration principle. The test gears are connected by shafts to a pair of helical gears that complete the loop. The torque is applied through a hydraulic loading mechanism that twists one slave gear relative to the shaft that supports it; therefore the torque is usually reported as a function of the hydraulic pressure. The drive motor only has to supply enough power to overcome the losses in the system. The test gears are lubricated with an independent oil system. The speed, torque, and input oil test temperatures can all be controlled.

The slave gears, since they operate at the same rotational speed, will have a gear mesh frequency that is coherent with the synchronous average. In short, the filtering advantage of the synchronous average will not reduce them in amplitude.

During health monitoring tests, an infrared optical sensor on the input shaft is used for the once per revolution signal. Typically, there are two accelerometers used for HUMS research, one mounted on the outside of the test housing, with the other mounted in the test section directly on the bearing cover plate.

The once per revolution tachometer signal is generated using an infrared optical sensor that is located on the input shaft to the test gearbox. The once per revolution infrared optical sensor detects a change in the reflectivity of an infrared light. The connecting shaft has a piece of highly reflective silver colored tape and is fastened with epoxy to the black oxide coated shaft. This provides a reliable signal that has good dynamic performance.

Two research accelerometers are mounted on the test gearbox. The first one (and only one for the first test) is located on the housing of the gearbox. In this paper, this accelerometer is referred to as the "Housing" accelerometer. The location was chosen based upon previous modal analysis testing on an identical gearbox. [4] It is piezoelectric with a frequency response from 20 Hz to 50 kHz. The second accelerometer is mounted 30 degrees clockwise from the vertical centerline for the right (driven) shaft on the bearing retention cap inside the gearbox. This one is referred to as the "Bearing Cap" accelerometer. It is also piezoelectric, but smaller and has a frequency range from 1 Hz to 30 kHz. The location is in the load zone of the bearing and provides the most direct transfer path for the vibration to travel. The configuration is shown in Figure 2. A third accelerometer, identified as the "Facility" accelerometer enables the tests to be run in an unattended mode and monitors the coarse vibration and provides a safety shutdown.

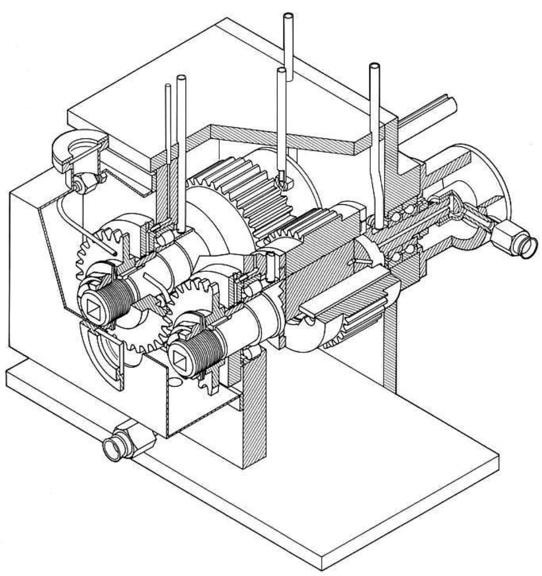

Figure 1. Spur Gear Test Facility

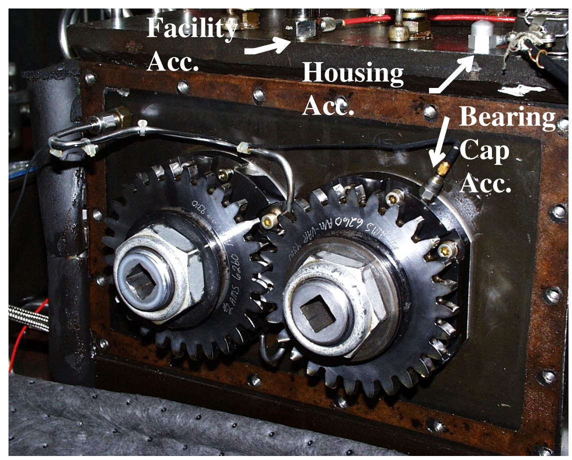

Figure 2. Accelerometer mounting locations with access cover removed

Gear descriptions: The test rig uses a pair of spur gears having 28 teeth, a pitch diameter of 88.9 mm (3.50 inch), and a face width of 6.35 mm (0.25 inch). During a surface fatigue test, the gear faces are offset by 2.8 mm (0.11 inch) to allow a higher surface stress without a corresponding increase in the bending stress. For bending fatigue tests, however, the gears are in contact across the full face width. The tests are also run at a higher torque than normal to assist in the formation of a crack front. A photograph of a test gear is shown in Figure 3.

The slave gears (Figure 4) have 35 teeth, a pitch diameter of 88.9 mm (3.50 inch), and a face width of 38.1 mm (1.5 inch). They are run at full face width. As such, they are lightly loaded and under certain conditions generate more vibration than the test gears.

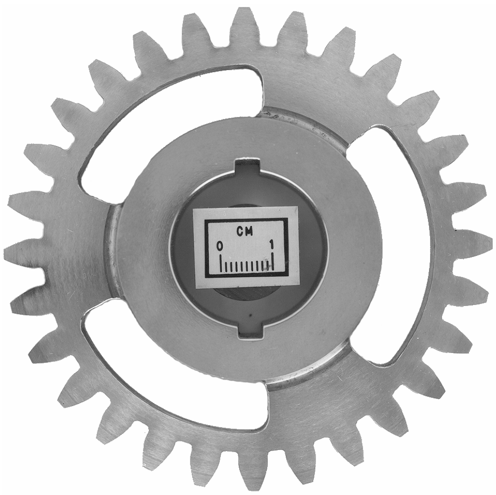

Figure 3. Sample gear for crack test

Figure 4. Slave Gear

Notch geometry: For bending fatigue tests, a notch is machined in the root area of the gear to provide a concentrated flaw from which a crack could initiate. This location is the point of highest tensile stress on the gear tooth root-fillet region. The higher stress provides the best opportunity for crack propagation.

The notch traverses the entire face width of the gear and is created using electrical discharge machining (EDM); this process is chosen for its ability to control the size of the notch. The size of the notch is controlled by both the shape and electric current of the electrode and is typically 0.254 mm (0.010 inch) deep.

Results: These crack tests are run at an overloaded condition to accelerate failure. It will be shown that it is difficult to determine crack initiation on these gears. It is desired to run the tests at overloaded conditions to initiate a crack, and then reduce the load to observe stable crack growth. This allows a more accurate study of the vibration signature during the critical crack growth period.

During the first test, only the housing accelerometer was used. The bearing cap accelerometer was installed between the first and second tests, and was available for the remainder of the tests.

Figure 5 illustrates the effect that other synchronous gear meshes can have on a vibration signal. In the uppermost plot, the synchronously averaged vibration is plotted as a function of angular position during one revolution. The middle plot shows the result of removing the slave gear mesh and its harmonics (not including those that are coincident with the test gear harmonics). Finally, the lower plot shows the contribution due to the slave gear mesh.

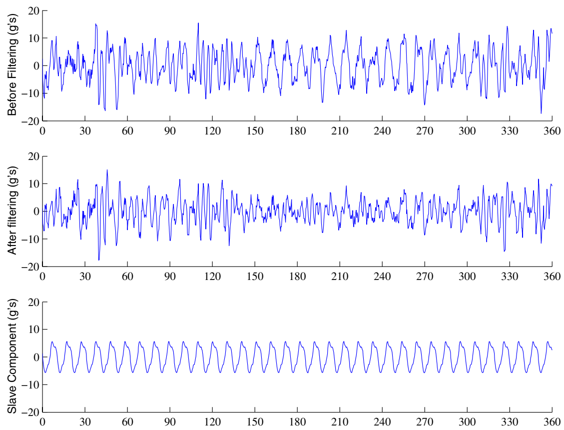

Figure 5. Effects of slave mesh filtering

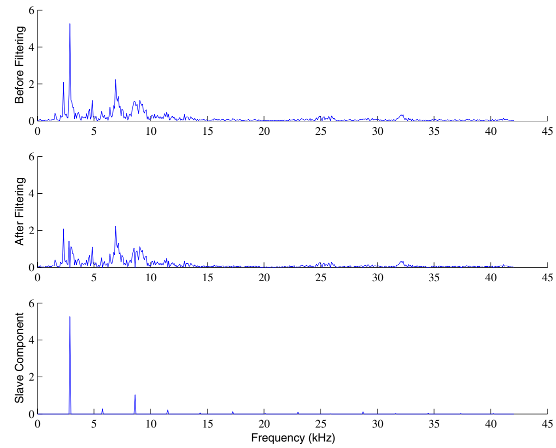

Figure 6. Fast Fourier Transform of effects of filtering

The frequency domain representation is presented in Figure 6. The plot on the bottom clearly indicates that all spectral lines associated with the slave gear mesh (with the exception of those that are coincident with the harmonics of the test gear) have been removed.

As mentioned previously, a total of 11 metrics were calculated for all of the vibration data. The volume of data can simply be described and illustrated with a few select charts. In general, most of the metrics were separated by a small offset between the unfiltered and the filtered signals. For most of the metrics, the results of the filtered metrics were larger in value.

Test 1: This test, run at 125-155 Nm (92-114 ft-lb) torque and 2500 rpm, produced a tooth fracture (Figure 7) after almost 237 hours. The original notch is readily visible in the fillet region on the left side of the gear tooth. The crack initiated at the edge of the notch and progressed to the fillet on the right.

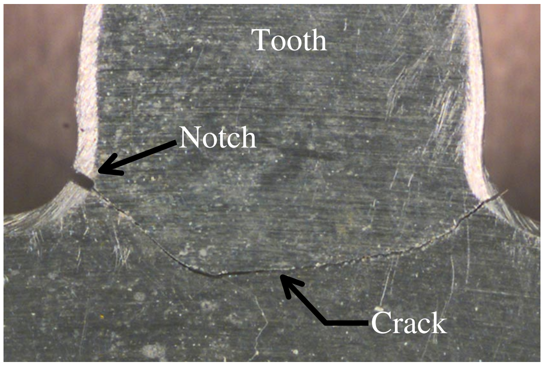

Figure 7. Gear Tooth fracture after Test 1

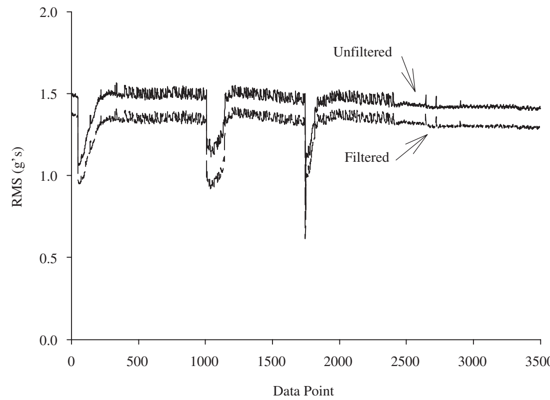

Figure 8. Test 1, Root Mean Square of synchronous average

Figure 8 shows the result of computing the RMS of the synchronous average. The upper line is the unfiltered synchronous average while the lower line results from filtering out the slave gear mesh. It makes sense for the filtered line to have a lower value as there is less information in that data set. This example shows that many of the metrics are not significantly affected by the slave gear mesh.

Figure 8 also shows the results of a premature shutdown of the facility (at approximately data point 1000), and an unexplained set of conditions at about data point 1700. Experience has shown that several of the diagnostic parameters take a significant amount of time to settle back into quasi-steady state conditions after an interruption. In this figure there is no obvious indication of crack initiation, progression or separation of the gear tooth.

Test 2: Test 2 was conducted at 5000 rpm and 155 Nm (114 ft-lb) torque. This test ended at 1.7 hours with a fracture through the rim (Figure 9) which may have been caused by either a corrosion pit resulting from incorrect storage of the gear or from operation at a resonance condition. At 1.25 hours, high vibration levels caused an automatic test shutdown. The gear was examined and a mark taken to be dirt or fuzz was noticed. This may have actually been the crack that eventually propagated through the rim.

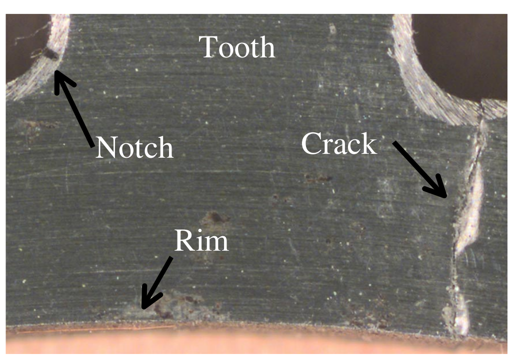

Figure 9. Gear rim fracture after Test 2

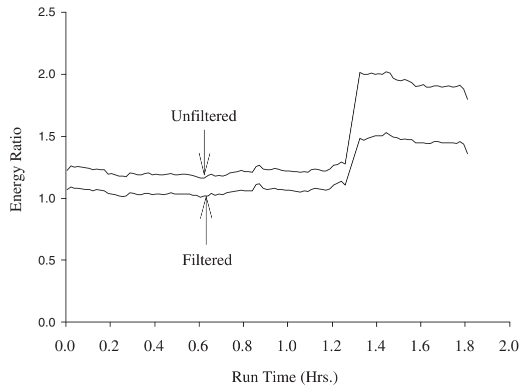

Figure 10. Test 2, Bearing Cap Accelerometer Energy Ratio

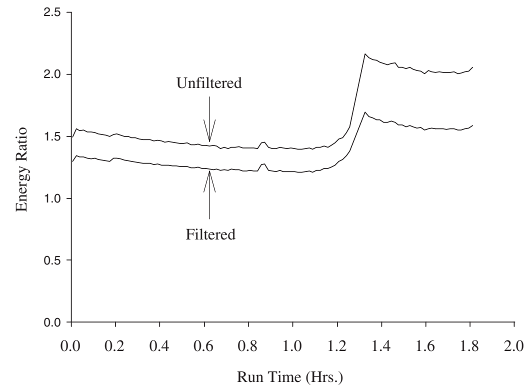

Figure 11. Test 2, Bearing Cap Accelerometer, Energy Ratio

Figure 10 and Figure 11 present the results of applying the Energy Ratio metric to the vibration recorded by the two accelerometers. In this test, almost all of the techniques examined indicate something at 1.25 hours. The combination of the shutdown and the damage combine to produce the peaks after 1.25 hours.

This metric exhibits some of the desired characteristics of an ideal metric. An ideal metric would show a step change at initiation of damage, a linear increase during damage progression with another step increase to a high level to indicate the loss of the tooth for the remainder of the run. At no time should a metric revert to a value that could possibly indicate a healthy condition when damage is present. The Energy Ratio shown in Figure 10 is preferred over Figure 11 since as time progresses (between 0 and 1.2 hours), the value of the metric does not decrease.

Test 3: This test also produced a fractured tooth (Figure 12). This fracture was not complete and progressed to approximately the tip of the arrow indicating the crack in the figure. The facility monitoring accelerometer detected a high vibration level due to the crack and shut down the system before the loss of the tooth. The test conditions were 4925 rpm at 115 Nm (84.9 ft-lb). The speed reduction was to avoid a resonance condition near 5000 rpm. The gear was later run at various torque settings until complete fracture occurred.

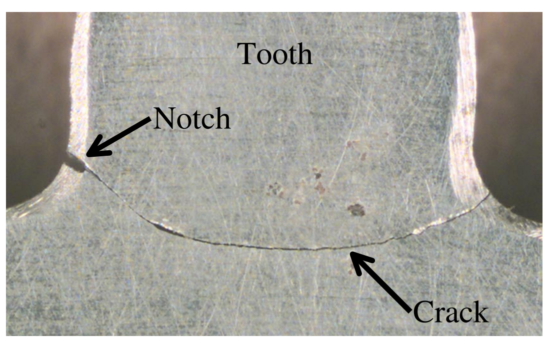

Figure 12. Gear tooth fracture after Test 3

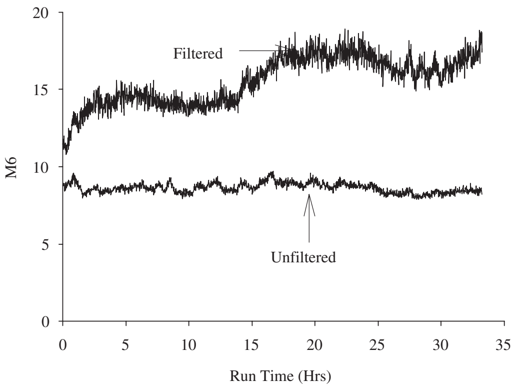

Figure 13. Test 3, Bearing Cap, M6A

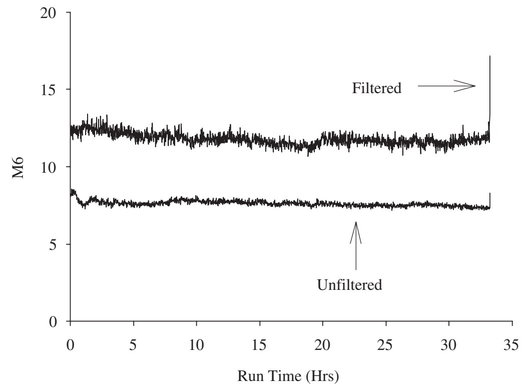

Figure 14. Test 3, Bearing Cap Accelerometer, M6A

Figure 13 does not appear to indicate any damage. On the other hand, Figure 14, with its sharp increase at the end, appears to indicate the damage about 1 minute before the rig shutdown. In this particular case, filtering out the slave gear mesh frequencies produced a profound effect on the metric. Since the filtered signal produced such a large increase in such a short time, there is more confidence that damage has been detected.

Conclusions: The tests conducted in this study reflect other previous experiments that show that no single processing procedure, metric, accelerometer mounting location routinely outperforms the others for gear crack detection. Several methods for feature extraction and detection appear to be required. At times, some failures are not reliably detected. This leads to several important conclusions that can be obtained from this testing:

1. For the commonly used vibration diagnostic parameters examined here, there is no single parameter that will reliably and accurately detect gear fractures until there is significant, possibly secondary damage (complete loss of tooth).
2. The techniques presented in this paper, while improving on existing techniques, still do not have sufficient robustness and accuracy.
3. Due to the high rotational speed in this study, using current techniques, it is almost impossible to be able to reliably detect a tooth fracture in sufficient time to be able to monitor its growth. The rotational speed should be reduced to allow the crack to propagate through several data acquisition cycles.

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Harry J. Decker

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13. ABSTRACT *(Maximum 200 words)*

Gear cracks are typically difficult to diagnose with sufficient time before catastrophic damage occurs. Significant damage must be present before algorithms appear to be able to detect the damage. Frequently there are multiple gear meshes on a single shaft. Since they are all synchronous with the shaft frequency, the commonly used synchronous averaging technique is ineffective in removing other gear mesh effects. Carefully applying a filter to these extraneous gear mesh frequencies can reduce the overall vibration signal and increase the accuracy of commonly used vibration metrics. The vibration signals from three seeded fault tests were analyzed using this filtering procedure. Both the filtered and unfiltered vibration signals were then analyzed using commonly used fault detection metrics and compared. The tests were conducted on aerospace quality spur gears in a test rig. The tests were conducted at speeds ranging from 2500 to 5000 revolutions per minute and torques from 184 to 228 percent of design load. The inability to detect these cracks with high confidence results from the high loading which is causing fast fracture as opposed to stable crack growth. The results indicate that these techniques do not currently produce an indication of damage that significantly exceeds experimental scatter.

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